TEST RIG FOR VALVES OF DIGITAL DISPLACEMENT MACHINES

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ABSTRACT

A test rig for the valves of digital displacement machines has been developed at Aalborg University. It is composed of a commercial radial piston machine, which has been modified to facilitate Digital Displacement operation for a single piston. Prototype valves have been optimized, designed and manufactured for testing and this paper presents examples of experimental results along with a thorough description of the test rig.

KEYWORDS: test rig, digital displacement technology, valve prototypes

1 INTRODUCTION

Traditional variable displacement fluid power machinery is known to be robust and energy dense, but also inefficient at lower displacement ratios. Digital Displacement Machines (DDMs) have been shown to achieve efficiencies above 95% for a wide range of displacement ratios [1] which makes fluid power much more attractive for several applications. DDMs achieve such high efficiencies even at low displacements since the power losses scales close to linearly with the machine displacement ratio opposed to traditional fluid power machines where the power losses typically scales more closely with the machine pressure [2]. DDMs consist of multiple parallel configured reciprocating pistons each connected to high and low pressure oil supply through two electrically controlled valves. By controlling the valves appropriately accordingly to the rotation of the shaft, pumping and motoring and idling cycles are possible. Idling cycles are obtained by keeping the low pressure valve open (and the high pressure valve closed) throughout the revolution effectively disabling the chamber. This operation mode involve low losses since the chamber is not pressurized which reduces friction and leakage losses between sliding surfaces. By varying the number of idling vs. active (pumping or motoring) cylinders on a stroke by stroke basis, the machine displacement can be controlled rapidly while maintaining high efficiencies.

The idea of such a digital variable displacement machine, often just referred to as digital pumps/motors, was conceived by a research group at the University of Edinburgh in the late 1980's. Later, the group founded Artemis Intelligent Power Ltd (AIP) which ever since has worked with maturing the technology. In 2010, AIP was acquired by Mitsubishi Heavy Industries (MHI) with the purpose of applying the AIP technology for wind turbines along with railroad- and ship-related products. In February 2015, MHI started testing of a 7 MW digital hydrostatic drive train installed in a wind turbine in Hunterston, England [3], [4], however no results with respect to performance have been published yet. Previously AIP have demonstrated efficiency improvements up to 27% when applying the Digital Displacement (DD) technology in an energy recovery system in hybrid commercial city bus [5] and have also applied the technology for a hydraulic excavator [6].

DDMs have also been studied by a few research groups in academia. Tampere University of Technology has published experimental results on digital hydraulic power management system (DHPMS) prototypes [7, 8, 9]. In [9] a six piston boxer pump is modified by replacing the original check-valves with actively controlled two-way on/off valves. Each cylinder was fitted with one low pressure valve and two high pressure valves connected to different loads. The valves are direct solenoid actuated prototypes and are capable of rapid switching (about 1 ms) and the pressure drop is 5 bar at 23 L/min (the DHPMS has a total displacement of 30 CC/rev, leading to a displacement volume of 5 CC per piston). A maximum efficiency of 85% in the speed range 500-1000 RPM and with approximately 100-150 bar machine pressure span. Efficiencies during partial displacement are reported in the range 30-80%. Plausible reasons for the decrease in efficiency is the excessive valve pressure losses, excessive leakage and mechanical friction. In [10] a DHMPMS is applied for direct boom actuation which shows significant potential for increasing the efficiency compared to the resistive methods typically used for cylinder control today.

A research team from Purdue University has also published experimental results on a single piston DD test setup [11] with the focus being investigation of different operating strategies [12] and efficiency [2]. Their approach is to use two stage bi-directional check valves as this enables operating strategies such as Partial Flow Limiting. The test-setup is based on a single piston driven by a symmetrical cylinder controlled by a servo valve to sinusoidally actuate the pistons. Limitations of the servo-valve bandwidth and available flow prohibit full stroke operation at higher speeds than 500 RPM, which leads to maximum flow rates of approximately 5 L/min. During testing the machine pressure span was approximately 30 bar. No results are presented with respect to valve performance or efficiency; additional details about the test stand is given in [13].

In [14] an experimental test-setup for a multi-cylinder DD machine is presented by the same group. The basis is a commercial in-line 3-piston check-valve plunger pump, fitted with commercial direct solenoid actuated normally closed poppet valves. The valves have switching times (incl. delay and transition time) of 8 ms, when boosted properly and a pressure drop of approximately 25 bar at 20 L/min [15] (corresponding to maximum flow rate for a 9.5 CC chamber operated at 700 RPM which is the highest tested speed) . Maximum efficiencies, at 500 RPM and 103 bar machine pressure span, of 90% were reported at full displacement for both motoring and pumping cycles. For partial displacement strokes, the efficiency, in worst case. dropped to approximately 60% at 30% displacement ratio.

The test rig presented in this paper is a single piston DDM, enabling testing DD operation with prototype valves. The basis of the setup is a modified five piston radial displacement motor where one of the chambers has been modified to facilitate DD oper-

Figure 1: Pictures showing the prototypes assembled (*right*) and disassembled (*left*).

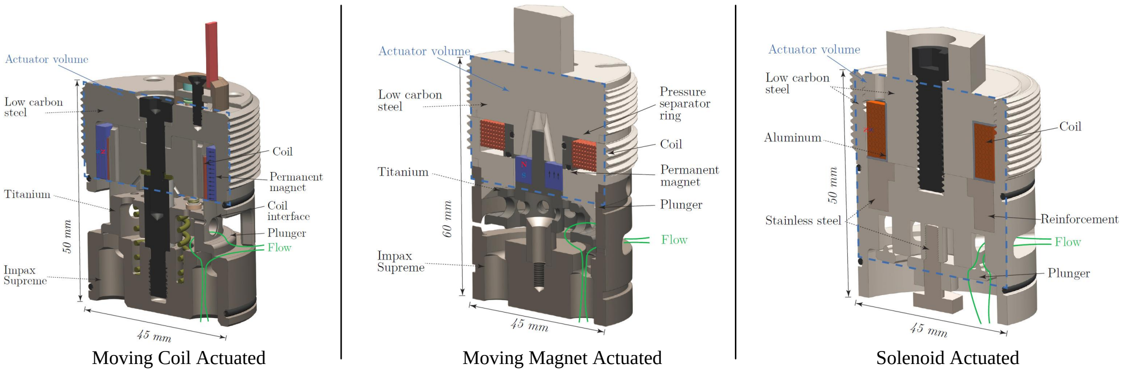

Figure 2: The prototypes shown in sectional CAD views.

ation. The chamber displacement of each cylinder is 50 CC/rev and when operated at the maximum rated speed of 800 RPM this leads to peaks flow of 125 l/min. The research objectives of the test rig are DD operating strategies and especially testing the performance, the durability, and the power losses of novel prototype valves designed in DD operation.

The following section presents some of the valve prototypes the test rig have been designed for testing. Then attention is turned towards the test rig where a detailed description of the hydraulics and the data acquisition and control system is given. Finally, some results obtained from the test rig are presented.

2 VALVE PROTOTYPES FOR TESTING

This section presents three different prototype (PT) valves for DDMs. All the valves are direct electromagnetically actuated normally open annular seat valves, but are actuated by different actuator topologies. One is moving coil actuated, one is moving magnet actuated valves and one is solenoid actuated. All the PT valves are shown in Figure 1 and Figure 2 shows sectional views from CAD models.

All of the PTs are cartridge valves with flow ports at the side and bottom of the valve. The outer diameters are identical for all PTs, and the O-ring grooves are placed identical, relative to the bottom of the valve for all PTs. This way, all the PTs can be installed in the same valve block. All the designs have been derived with the aid of multi-objective optimization and detailed information about each design is published in [16, 17, 18]. All valves are designed for operating a cylinder of 50 CC running at 800-1000 RPM with a machine pressure span of 350 bar. Each valve can function as either high or low pressure valve, however, since the PTs are normally opened valves they are arguably best suited as LPV. In case of actuator failure when using a normally open valve as HPV, the pressure chamber enters a state of high-pressure idling which is undesired as losses are increased [19].

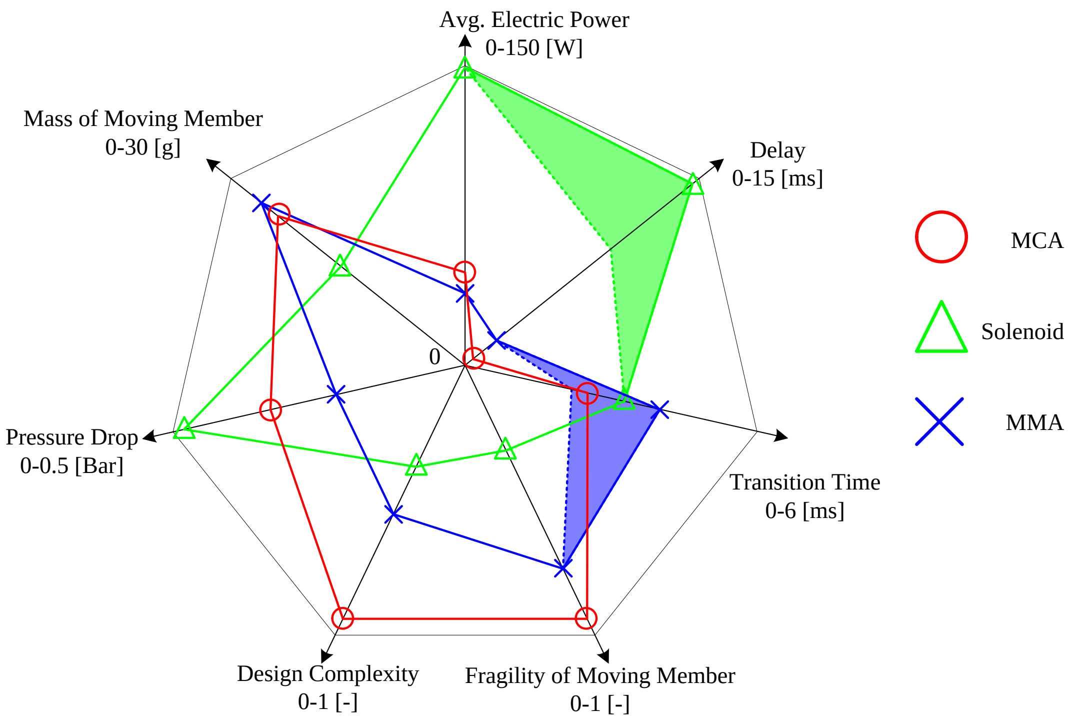

Figure 3: Radar plot comparing the capabilities of the three different prototypes.

A radar plot, comparing the capabilities of all three PTs, is shown in Figure 3. In total, seven criteria are used for the radar plot and common for all is that a low value is desired. The average electric power consumption is based on DD operation at 800 RPM[1]. The delay is defined as time from the voltage step is given until movement is initiated and the transition time is from movement is initiated until impact with the mechanical end-stop at the closed position. The given delays and transition times are based on standalone test of the valve switching performance where the displacement of the moving member was measured. The pressure drops have been measured at approximately ±125 L/min, corresponding to the maximum flow rate for a 50 CC cylinder at 800 RPM. Due to the design of the valves, the magnitude of the pressure drop depends on the direction of the flow. The pressure drop shown in Figure 3 is the average of the absolute pressure drops at the the maximum (positive) and minimum (negative) flow rate. The design complexity and the fragility of moving member criteria are assessed by the authors. The translucent areas of the figure indicate that certain criteria values may vary. For instance, the translucent red area indicates that the delay of the Solenoid actuated valve could be less, if boosting the current (very common technique for accelerating the current and flux build up). The translucent area for the MMA valve is included since the transition time is more dependent on the flow rate because no spring is included in this design. Instead, the MMA valve relies on magnetic latching towards the opened position.

The pressure drop is seen to be below 0.5 bar for all designs enabling high efficiencies. The pressure drop of the MMA valve is seen to be the lowest, this is primarily because a larger stroke length is used compared against the other PTs. The electric power consumption of the Solenoid actuated valve is seen to be significantly larger than for the MCA and

[1]At the moment, operation at 800 RPM have not been performed using the solenoid valve, hence this value is estimated based on standalone switching tests

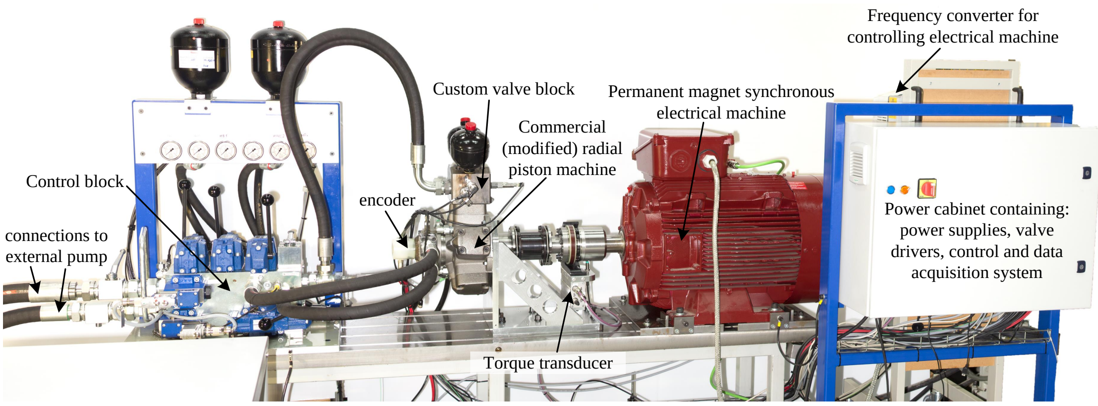

Figure 4: Picture showing the test rig.

MMA PT valves. This is primarily because the of the much larger time delay, occurring due to eddy current effects, during which significant electric energy is consumed. The design complexity and the fragility of the moving member of MCA PT is assessed much higher than for the other PTs. The coil of the MCA is attached to the moving member and the electric connection from the stationary part to the moving member complicates the design and is an issue of concern wrt. durabillity making the design more fragile.

3 TEST-RIG

The test rig is shown in Figure 4. The basis of the setup is a commercial radial piston machine, which have been modified to facilitate DD operation for one of its five cylinders. The radial piston machine is connected to a permanent magnet synchronous electrical machine (PMSM) which is used to control the rotational speed by means of an 90 kW ABB four-quadrant frequency converter. The radial piston machine is supplied using the hydraulic control valve block which again is supplied by a 250 kW variable displacement pump station (not shown in picture). The power cabinet contain data acquisition and control systems and valve driver.

Figure 5 shows a CAD model of the modified radial piston machine and the valve block with the MCA and MMA installed as LPV and HPV, respectively. The cylinder head is modified by blocking the original connection from the chamber to the port plate and by making a new connection through its top to the custom made valve block situated on top. The translucent green areas of the figure illustrate the cylinder volume while the red and blue translucent areas indicate high and low pressure respectively. The commercial radial piston machine is a Calzoni motor MR 250 [20]. This machine was chosen as basis of the setup since it has a relatively large chamber displacement volume (250 CC/rev) and can run at relatively high speeds (rated to 800 RPM) leading to peak flows of approximately 125 l/min for each of the pistons. Also, the radial configuration entails sufficient space for installation of PT valves. Finally, the machine has high values of mechanical and volumetric efficiency as a consequence of the employed mechanism for converting the rotation of the shaft to translation of the pistons i.e. internal piston support and a telescopic piston-cylinder mechanism [21][2].

[2]The first machines of AIP also uses the telescopic piston/cylinder mechanism [22]. The 3.5 MW MHI DDM uses a crank-slider mechanism [23].

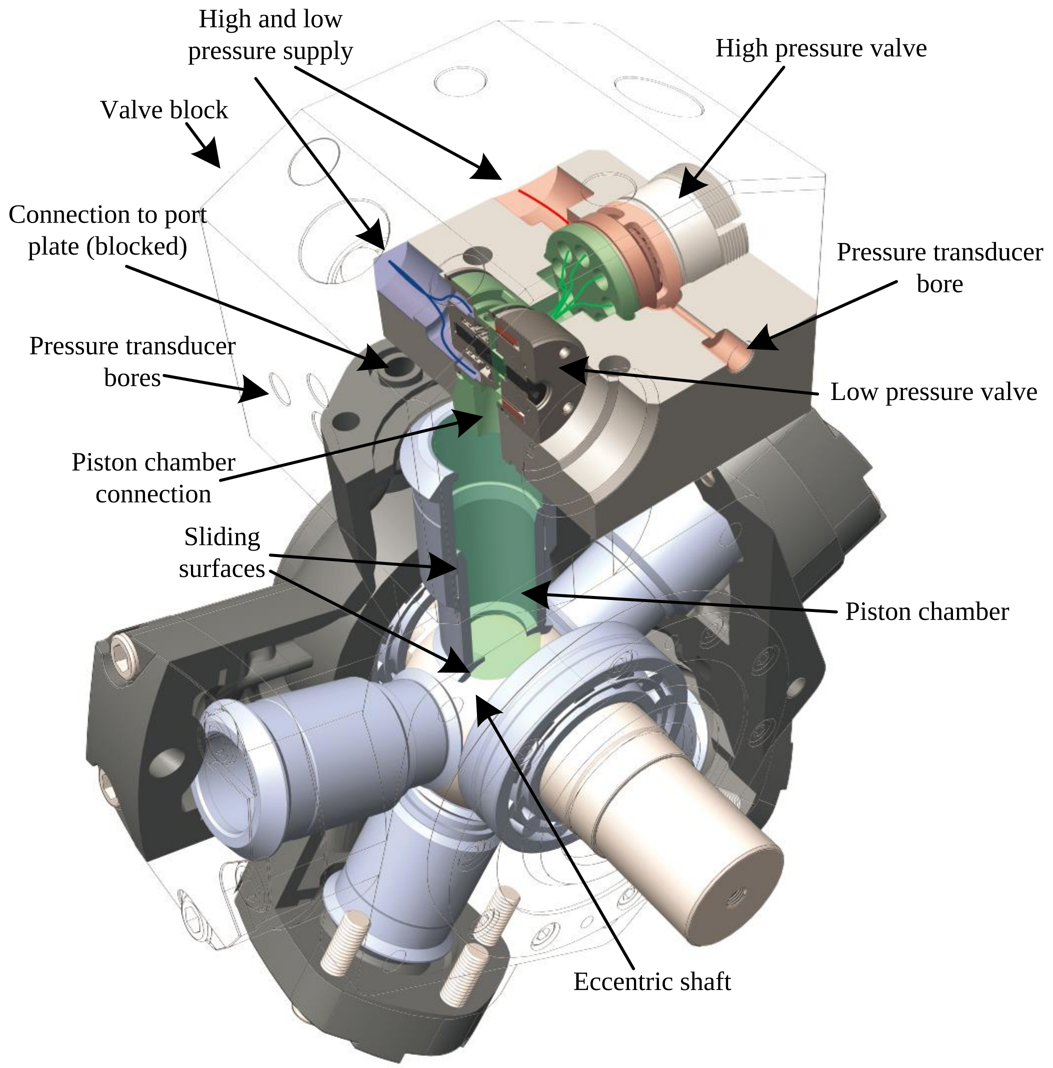

Figure 5: CAD model showing the modified machine and the custom valve block equipped with the MCA and MMA PT valves.

The following subsections describes the hydraulics and the data acquisition and control system in more detail.

3.1 FLUID-POWER

The hydraulic circuit layout of the test rig is shown in Figure 6, including all valves, accumulators and pressure sensors. The control block is designed to facilitate pumping, motoring and idling DD operating cycles while also supplying appropriate pressures and flows for case flushing and the remaining (unmodified) pistons of the machine. An external 250 kW variable displacement axial pump is used to supply the control block. Several accumulators are used to stabilize pressure lines which are prone to oscillation due to the nature of DD operation. The pressure reducing valves (1,3) are used to control the pressures of the high pressure line p_{H} and the intermediate pressure line $p_{\mathrm{int,reg}}$. The intermediate pressure line is used supply the unmodified four pistons of the machine and for flushing of the machine casing using a flow control valve (9). While testing DD oper-

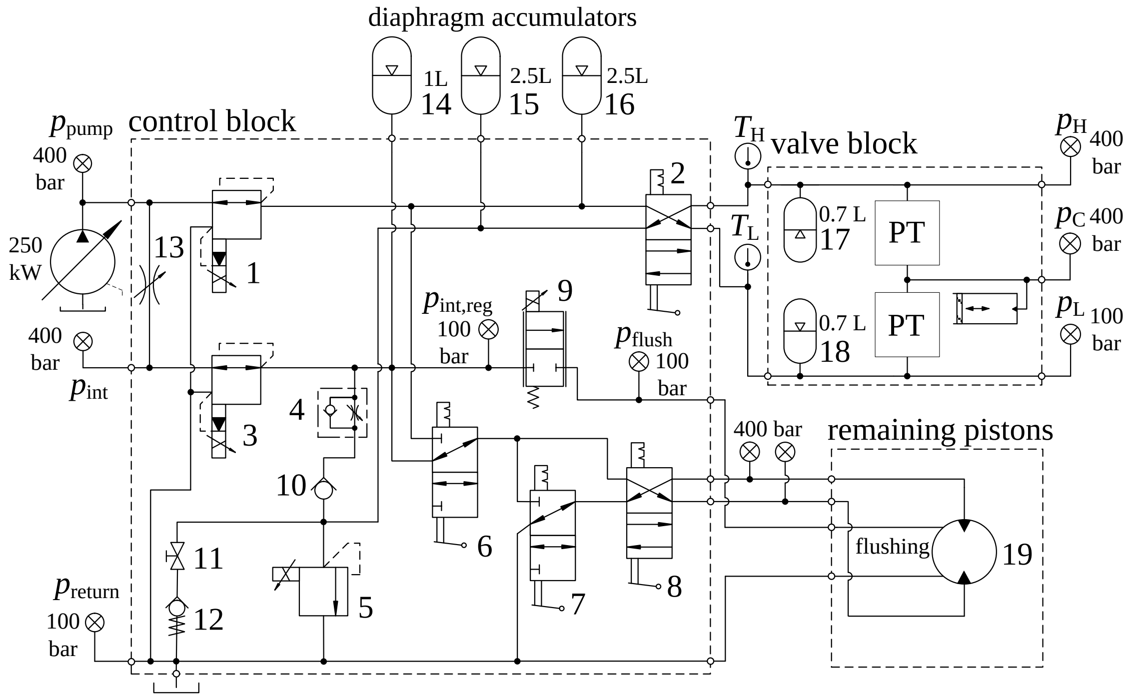

Figure 6: Hydraulic circuit layout of the test rig.

ation, the remaining pistons are short circuited by setting the 3/2 directional control valve (7) opposite of drawn. When performing DD motoring cycles the 4/2 directional control valve (2) is set opposite of drawn, the gate valve (11) is open and the pressure relief valve (5) is set to the high pressure supply level. In case of the DD pumping cycles the gate valve (11) is closed, the pressure relief valve (5) is used to control the pumping pressure and dissipate the hydraulic power and the 4/2 directional control valve (2) is set as drawn. The purpose of the flow control valve (4) and check valve (10) is for startup in pumping operation. Note that the 2.5 L accumulators are implemented with shut-off blocks, not shown in 6.

3.2 Instrumentation

As a test-rig for novel valve designs, it is required to measure as many parameters as possible to both verify models, and aid in the design process. Many sensors are also used in the control system, for monitoring, and for safety purposes. In the following section the measured parameters, and sensors used, will be presented.

Pressure Measurements

The placement and range of each pressure sensor is shown in Figure 6. The sensors used are Rexroth HM20, which are in accuracy class 0.5 and have a settling time <1 ms. In order to capture the rapid pressure changes, during pressurization and depressurization in the chamber, a Kistler 6005 piezoelectric pressure sensor is used in combination with a Rexroth HM20 sensor. The Kistler sensor is only capable of measuring dynamic pressures, so a low pass filtered signal from the Rexroth sensor is combined digitally with the high pass filtered signal from the Kistler sensor to create the merged signal.

Temperature

Keeping track of the oil temperature is essential as it influences the viscosity of the oil and overheating of the oil may damage it. The temperature is measured, using HydroTechnik HySense TE sensors, immediately up- and down-stream of the the valve block as shown in Figure 6. The temperature of the radial piston machine case is also measured using a thermocouple.

Torque, Position, and Speed

The shaft position is measured using a an incremental Scancon encoder with 5000 ppr, and the shaft rotational speed is calculated based on the ticks per time unit of the encoder signal. The torque on the shaft connecting the radial piston machine and the PMSM is measured using an HBM T12 inductive torque transducer. The ABB frequency converter has a dedicated encoder from which the shaft position and speed also is calculated. The torque is also estimated by the ABB frequency converter based on measurements of the phase currents.

3.3 Control & Data Acquisition System

Figure 7 illustrates the basic architecture of the control and data acquisitions system. A Windows PC, running National Instruments LabVIEW, is used as human machine interface (HMI). The main controller is a National Instruments cRIO 9039 which features both a real-time processor and an FPGA. The valve driver board constitutes a National Instruments MyRIO 1950, a custom made interface and data acquisition board and an amplifier board based on a Texas Instrument DRV8412 dual full bridge PWM motor driver. The interface board carries out voltage and current measurements in order to control the actuators (both voltage and current control is possible). Also, the DC supply voltage of the valve driver board is measured. A eight-channel 16 bit DAC with 200 kS/s per channel is used. The valve drive board has two channels each rated at 52 V and 24 A peak (14 A continuous). When the respective valves are inactive, the PWM driver output is high-Z stated; it is possible to measure the actuator back electro-motive force in this state.

The communication between the main controller and the valve driver board is deterministic ensuring that measurements and control signals are synchronized. The main controller functions as a state machine and handles the control of all valves and the PMSM (incl. freq. converter). Also, the main controller collects and buffers all measured signals and forwards all data to the HMI for visualization. The HMI is used to change the machine settings, for real-time monitoring, and for saving of data to disk.

4 EXAMPLES OF RESULTS

Figures 8 and 9 show DD motoring and pumping operations, respectively, both at 800 RPM and 100 bar machine pressure span.

During the measurements, the MMA valve was used as HPV and the MCA valve used as LPV. For the presented measurements, the 2.5 L accumulators connected to the high and low pressure line (15 & 16 cf. Fig. 6) were shut off and the 0.7 L high- and low-

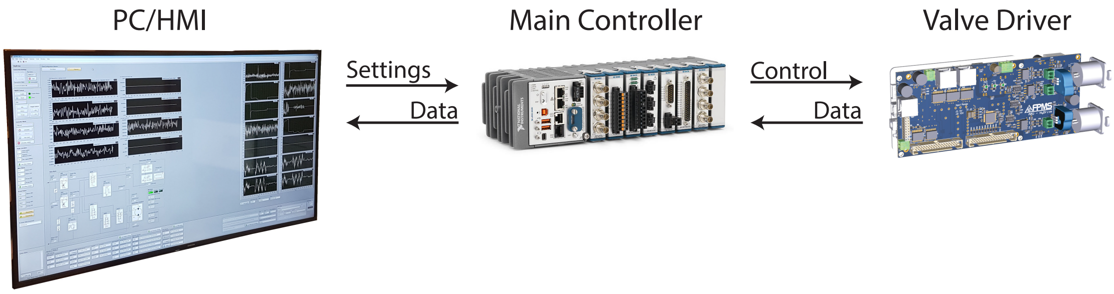

Figure 7: Illustration of the control and data acquisition system architecture.

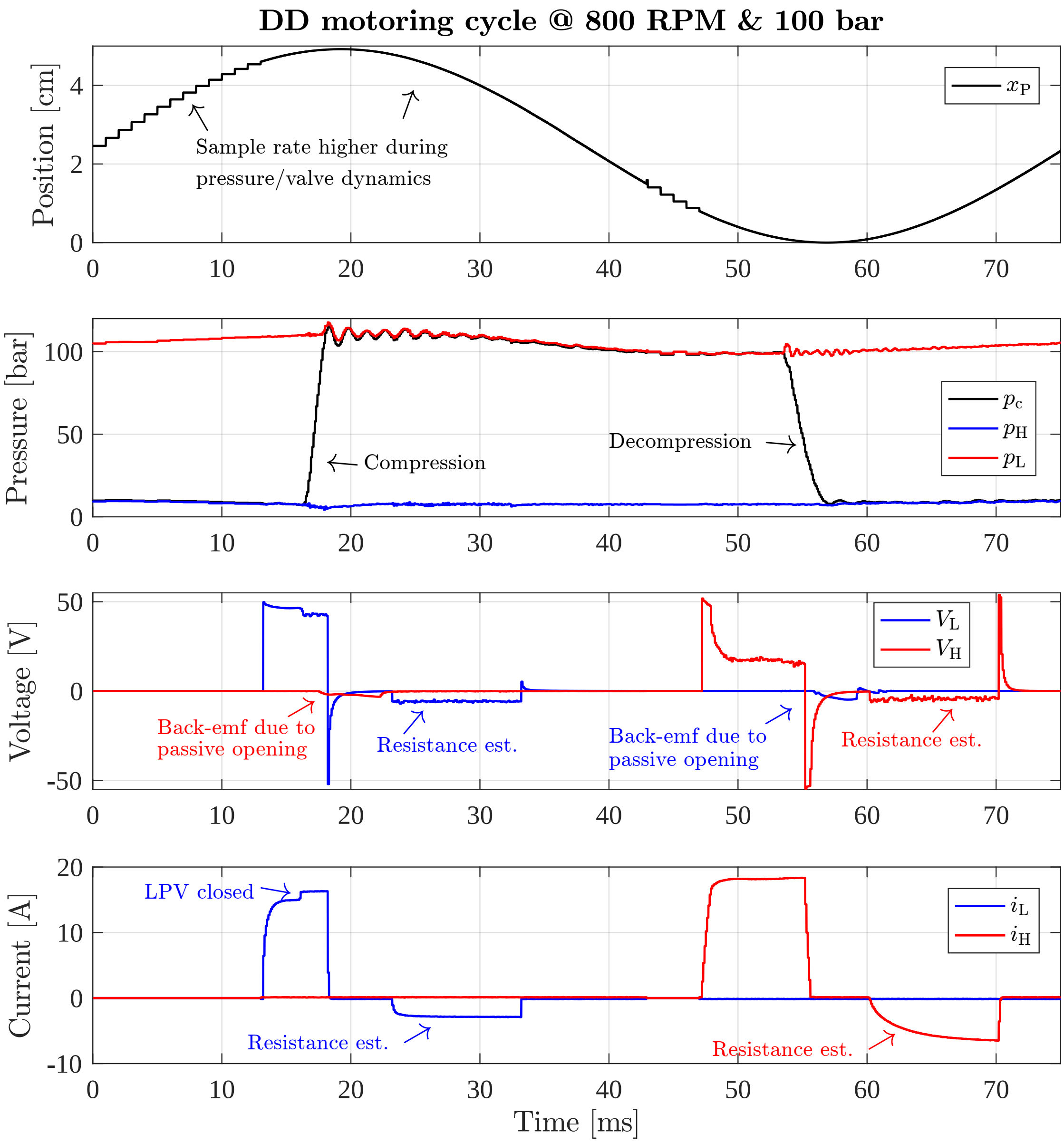

Figure 8: Example of DD motoring at 800 RPM and 100 bar machine pressure span using the MMA valve as HPV and the MCA valve as LPV. During measurement the oil temperature was 40°C.

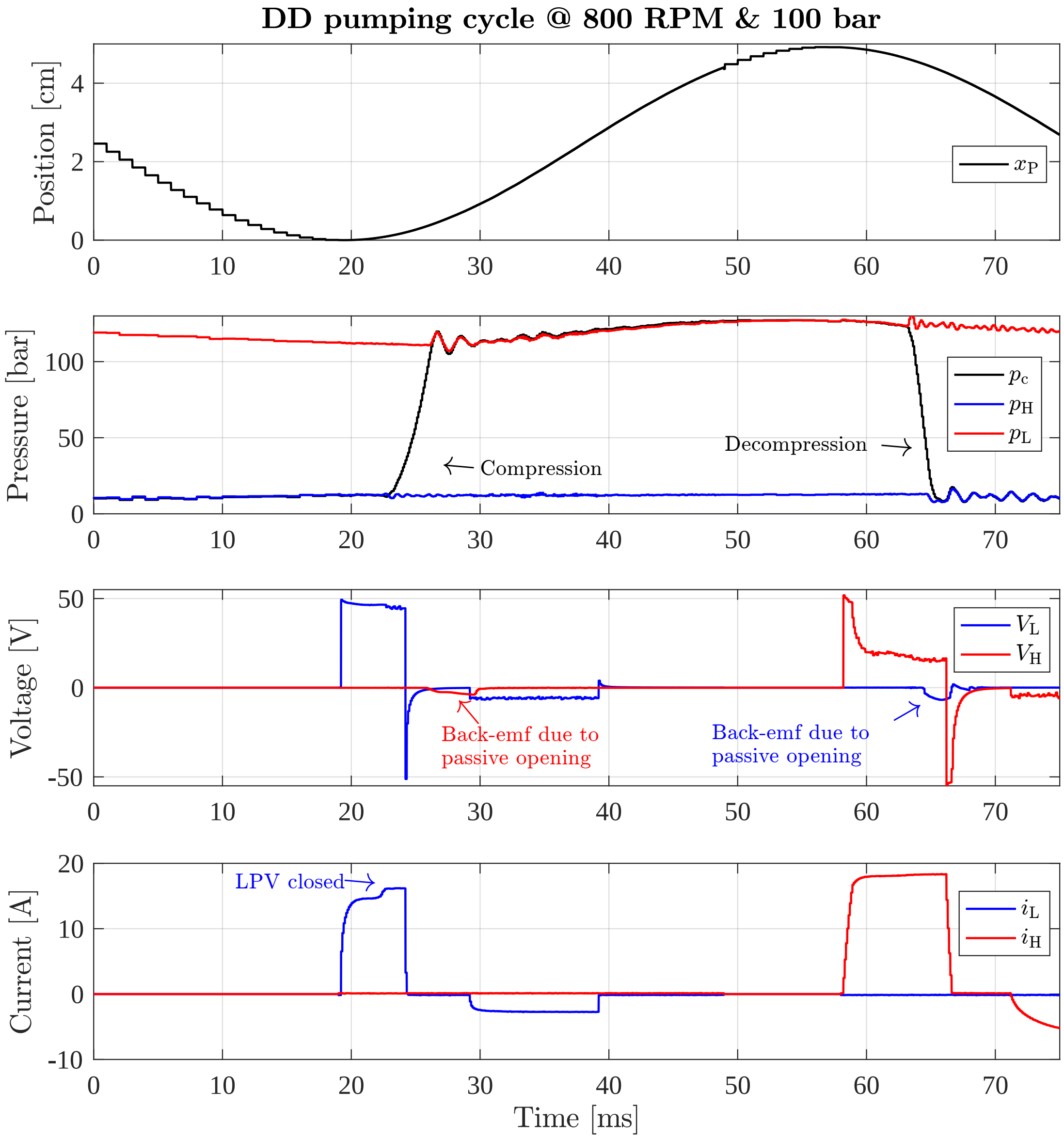

Figure 9: Example of DD pumping at 800 RPM and 100 bar machine pressure span using the MMA valve as HPV and the MCA valve as LPV. During measurement the oil temperature was 47°C.

pressure accumulators of the valve block was pre-charged to approximately 50 bar[3] and 5 bar, respectively. The measurements have been sampled at 100 kHz during valve/pressure dynamics and at only 1 kHz at times with no valve/pressure dynamics. This is easily visible from the position x_p which is calculated based on encoder signal. For the DD motoring cycle, the compression and decompression are completed relatively closely to the respective dead centers, leading to maximum displacement and valve switching's at the lowest possible flow rates. For the shown DD pumping operation cycle, the compression/decompression occurs later than the respective dead centers indicating the valves could have been closed earlier resulting in increased displacement. Considering the volt-

[3]A pre-charge pressure of 50 bar is considered too low for the 105-115 bar present at the high pressure line during measurements. It is expected that the ripple of the high pressure line can be attenuated further by increasing the pre-charge pressure.

ages and current of the actuators, the waveforms reveal useful information about the behavior of the valves. The increase in current i_L of the MCA valve 3-4 ms after the voltage signal was given indicates that the valve is closed as the velocity dependent back-emf voltage goes to zero leaving additional voltage to drive the current. Based on this, the valve closing times (incl. delay which is very small for moving coil actuators) are 3.1 ms and 3.3 ms for the motoring and pumping cycle respectively. During the passive openings of the LPV, occurring after the decompressions, the actuator coil is open circuited while the movement induced back-emf voltage is measured. The induced voltage during the passive opening of the LPV is seen to be larger in magnitude for the pumping case. This corresponds well with the opening of the LPV being performed at larger flow rates for the pumping case since this makes the valve open faster. Based on the duration of the back-emf pulse the valve opening times are estimated to be 3.3 ms and 2.2 ms for the motoring and pumping case respectively. The closing times of the MMA valve are not directly interpretable from the current forms i_H and are instead estimated as the time from the voltage step is given until the decompression of the chamber initiates. Using this method the closing times are estimated to be 6.2 ms and 5.1 ms for the motoring and pumping case respectively which also corresponds well with the closing of the HPV being performed at a larger flow rate for the pumping case. The opening times of the MMA valve are not easily interpretable from the current, voltage or pressure waveforms and have not been estimated. Subsequent to the active closings of the valves, a small voltage is given to the coil which is intended for resistance estimation purposes. The electric power consumed by the actuators has been calculated by integrating the current and voltage product over the duration of the entire operation cycles. For the MCA valve, the consumed power was approximately 3.5 J for both cases, corresponding to approximately 0.8% of the transferred power. The MMA valve consumed approximately 2.7 J in both cases which corresponds to 0.6% of the transferred power. It is expected than the electrical power consumption can be reduced by up to 40% by tuning the duration of the closing signals.

5 CONCLUSIONS AND FURTHER WORK

The presented test rig enables testing of the operation cycles of Digital Displacement Machines using prototype valves. At this moment, motoring and pumping digital displacement cycles have been tested up to 800 RPM and 100 bar machine pressure span. The immediate experience with the test-rig is that it possess the necessary features and instrumentation for thorough testing of DD valve prototypes, operating strategies etc. Further work includes operation at machine pressure spans up to 350 bar which will increase the demands to several components of the set-up and in particular the prototype valve under testing.

6 Acknowledgements

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